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NASA CR 137,787

Final Report

OXYGEN BOOST PUMP STUDY

Report 75-12264

December 1975

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AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
DIVISION OF THE GARRETT CORPORATION

Prepared Under Contract No. NAS2-8794

for

AMES RESEARCH CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

GARRETT

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

FOREWORD

This is the final report for the Oxygen Boost Pump Study Program. This program was conducted by the AiResearch Manufacturing Company for the NASA-Ames Research Center under Contract NAS2-8794. The program was initiated in April, 1975 and was completed in December, 1975.

A breadboard boost pump, built and tested under a prior study program described in AiResearch Report 74-410521A, dated 4-15-75, demonstrated the free piston expansion concept, generated performance data as a baseline for evaluation and comparison of follow-on designs, and indicated areas where improvement was required. This program was performed for NASA Crew Systems Division, Johnson Space Center, Houston, Texas, under ALSA Program Task Order 186 of Contract NAS9-10465.

The program described herein utilized the data from the above mentioned-program to define an oxygen boost pump that is simpler to manufacture and requires less motor gas (gas vented into the atmosphere) than the previous design. Special thanks are due to the Contract Technical Monitor, Mr. Bruce Webbon of the NASA-Ames Research Center.

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I. INTRODUCTION

It is likely that extravehicular activity (EVA) will be used for future manned spaceflight programs on a routine basis to perform such activities as inspection, servicing, and repair of the vehicle and experiments. In order to provide a minimum weight and volume EVA system, it is necessary to use a high pressure oxygen storage system. Because these systems are designed for multiple use during a flight, a method must be obtained to recharge these high pressure oxygen tanks from the spacecraft oxygen supply. Normally, oxygen is stored in the spacecraft in supercritical oxygen tanks, having an operating pressure of +06 6.205 Pa (900 psia). The oxygen boost pump described in this report is a simple, reliable device, which can be used to charge the high pressure oxygen tank in the EVA equipment from spacecraft supply. The only interface with the spacecraft is the +06 6.205 Pa supply line.

This report presents results of study program activities for the oxygen boost pump, conducted by AiResearch for the Ames Research Center, Moffett Field, California, under Contract NAS 2-8794.

This report summarizes the breadboard study results and oxygen tank survey, (documented in the mid-term Report, AiResearch Report No. 75-11603) and presents the results of the flight-type prototype design and analysis.

2. SUMMARY

2.1 BACKGROUND

A breadboard compressor, built and tested under a prior study program described in AiResearch Engineering Report 74-41052A, dated April 15, 1975, demonstrated the free piston expansion concept, generated performance data as a baseline for evaluation and comparison of follow-on designs, and indicated areas where improvement was required. This study program was performed for NASA Crew Systems Division, Johnson Space Center, Houston, Texas.

The midterm report 75-11603, dated June 26, 1975, documented the study effort to that date. This included 1) additional analysis, 2) an update of the computer program with improved modeling to reflect actual performance of the compressor and to predict performance of the flight-type prototype, and 3) design innovations for the flight-type prototype to improve energy management. These design innovations include a redesign of the automatic cycle valve, capacitor, capacitor scheduler, and control orifices. The study to this point also recommended that final compressor output should be limited to +07 2.758 Pa (4000 psia) maximum.

In addition, a preliminary evaluation of compression thermal effects indicate that sufficient heat dissipation will be inherent to the compressor design so that the compressed gas temperature will not exceed +02 3.442 t_K (160°F), when operating in cabin ambient temperatures up to +02 2.998 t_K (80°F).

This final report proposes continuance of this program, resulting in the detail design, fabrication, and testing of a flight-type prototype.

2.2 ASSUMPTIONS - DESIGN CRITERION

An inflow control valve will be used to control pumping frequency, and an adjustable tank pressure sensing electrical switch will be used to control tank pressure.

The following performance parameters were utilized in the modeling analysis and design of the flight weight prototype:

Compressor discharge pressure design point:	+07 2.758 Pa (4000 psia)
Compressor discharge pressure range:	+06 6.205 Pa to +07 2.758 Pa (900 to 4000 psia)
Supply pressure design point:	+06 6.205 Pa (900 psia)
Supply pressure range:	+05 6.895 to +07 1.379 Pa (100 to 2000 psia)
Cabin flow design point:	+01 7.560 kg/s (1.25 lbm/h)

NOTE: Cabin flow (motor vent flow plus leakage) is equal to the total flow to the compressor minus compressor discharge flow.

2.3 OPERATION OF PROPOSED FLIGHT-TYPE PUMP

The proposed flight-type prototype pump, shown on Figure 2-1, operates in the following manner:

The spacecraft oxygen supply +06 6.204 Pa (900 psia) enters the pump at the oxygen inlet port. Oxygen will flow through the compressor inlet and outlet check valves and into the oxygen tank, without any pumping action, until the oxygen tank is precharged to the supply pressure. This inlet supply oxygen is also being directed through the inflow control valve into the gas capacitor. When the force developed by increasing capacitor pressure, acting over the trigger area on the motor piston, approaches supply pressure acting on the compressor piston area, the trigger valve starts to leak, exposing motor piston area to capacitor pressure, causing the motor piston to move. Initial movement fully uncovers the trigger port, which exposes the full motor piston area to capacitor pressure, which rapidly accelerates the motor piston, providing the pumping energy. As this assembly (motor and compressor pistons) moves toward the compressor end of the pump, the compressor inlet check valve closes, trapping the gas in the compressor chamber. This gas is then compressed and flows out to charge the oxygen tank. At the end of the compression stroke, the motor gas is vented to cabin through vent holes in the side of the unit. The supply pressure of +06 6.204 Pa (900 psia), entering the compression chamber through the inlet check valve, provides the motive force for the return, or suction stroke.

The bellows assembly, located above the capacitor, schedules the position of the capacitor piston to increase the capacitor volume as tank pressure increases. This varies the quantity of motor gas as a function of oxygen tank pressure, which matches the motor energy to the required compressor energy, so that motor gas is not wasted at lower tank pressures.

The flight-type prototype pump differs from prior version in the following aspects:

a. <u>The motor capacitor and triggering valve have been improved</u>. A differential area bellows and spring assembly senses increasing tank pressure to increase capacitor volume. This increases motor energy per cycle to match pumping energy requirements. The triggering mechanism is separate from the energy scheduling mechanism. The triggering device initiates the cycle when proper energy charge has accumulated in the capacitor, and discharges to the motor through an enlarged flow area to minimize flow losses. Unlike the prior automatic cycling valve, the new capacitor and triggering mechanism permits operation over a wide range of supply pressures.

b. <u>A vernier adjustable inflow control valve has been added</u>. The valve allows adjustment of capacitor charge rate and, therefore, pumping speed and cabin flow rate. Capacitor charging time can be reduced and pumping frequency and vent flow to cabin can be increased.

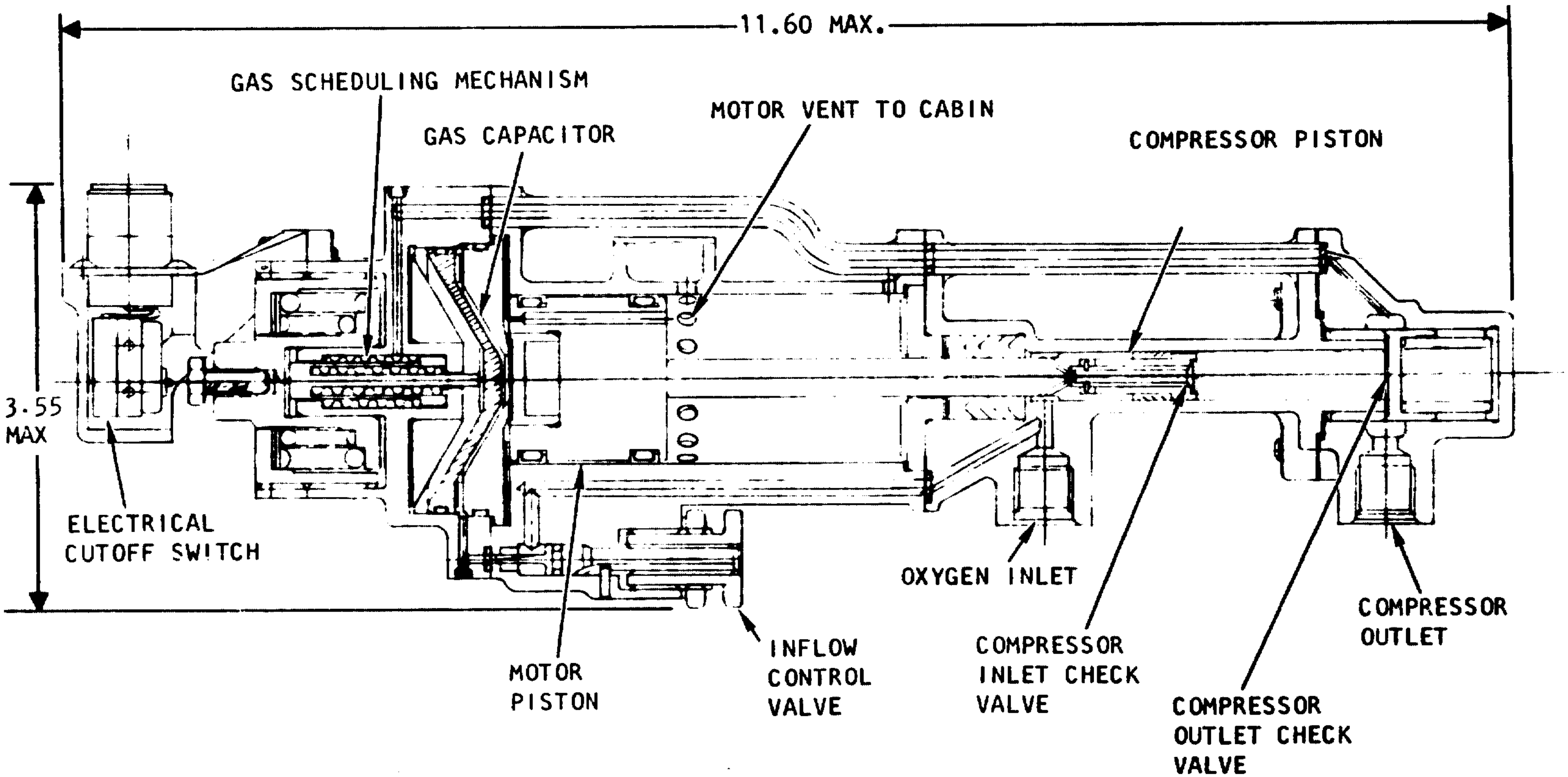

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Figure 2-1. Proposed Flight-Type Protype Pump Design

c. A pressure indicating electrical switch has been added. The switch is actuated by an adjustable screw actuator mounted to the tank pressure sensing bellows assembly. The switch can be used to shut off supply pressure to stop pumping at a preselected pressure. A snap-acting, pneumatically-actuated valve would be used instead of the electrical switch in an actual flight application to avoid an electrical interface with the spacecraft.

d. The variable volume capacitor piston has increased area and has been conformed to minimum residual volume when cycled. The increased area reduces the stroke required from the bellows volume scheduling device over the range of operation. The reduced residual or dead capacitor volume provides a small but significant increase in available pumping energy per unit of charge.

2.4 SPECIAL FEATURES

The design selected for the flight weight prototype incorporates many special features, which are described below.

a. Pumping over a range of supply pressure. The pumping process is dependent on triggering and stroking. The triggering process is independent of supply pressure since the force balance between the trigger and compressor piston at the start of stroke is a function of their respective areas, which do not change. The trigger area is slightly greater than the compressor piston effective area, and they are both fed by a common source of supply pressure. Triggering occurs when capacitor pressure reaches a value somewhat below compressor cylinder pressure, independent of level.

   The stroking process varies considerably with variations in supply pressure. As supply pressure goes down, pumping energy requirements increase (increased compression ratios) and stored capacitor energy decreases. Trigger cycling continues, but the pumping energy unbalance causes short stroking. At increased supply pressures, excessive energy is stored in the capacitor, resulting in high motor gas usage and impact of the motor piston at the end of the stroke.

   The pump will continue to operate over relatively large variations in supply pressure, but gas consumption efficiency will degrade and service life may suffer.

b. Conservation of motor gas-pumping efficiency. Expansion of motor gas ranges from 15:1 to 5:1 at peak pump output. Although useful pumping energy is not extractable over the entire stroke, this expansion reduces the motor gas requirements and greatly improves pumping efficiency.

c. Automatic adjustment of motor energy. Discharge pressure feedback to the capacitor scheduling device adjusts the capacitor volume to match pumping requirements. This prevents excessive energy overstroking at low output pressures, short stroking at high output pressures, and maintains peak pumping efficiency over the entire pumping range.

d. Preselection of tank pressure and automatic cutout. Discharge pressure feedback to the capacitor scheduling device (bellows assembly) provides a linear mechanical stroke proportional to tank pressure. A screw adjustment on this mechanical output is used to actuate an electrical switch for setting a predetermined pressure level. The electrical signal, when switched, can be used directly or through a latching relay to interrupt supply pressure, which will stop the pump.

e. Automatic initial charge of pressure vessel to supply pressure. The compressor inlet and outlet check valves allow the vessel being charged to be filled to supply pressure automatically. Compared to other positive displacement devices that do not allow flowthrough, considerable pumping gas is conserved at the design point supply pressure of +06 6.205 Pa (900 psia). Devices without flowthrough would require additional bypass components to achieve the same results.

f. A maximum pump discharge pressure of +07 2.758 Pa (4000 psia). This pressure was selected based on overall system considerations, which included tank-gas compressibility relationships, other component complexities, and pump design and reliability impacts. In the pump design, this is reflected in several areas: 1) maximum capacitor volume--reset mechanism stroke at +07 2.758 Pa tank pressure; 2) compressor seal design; 3) compressor cylinder, inlet and outlet check valve design; and 4) cutout switch design.

g. Pumping at very low rates to match cabin flow requirements with optional selectable increased flow rate. Inflow to the capacitor determines charge time per cycle and therefore cycle rate or frequency. The vernier adjustable capacitor metering valve allows capacitor charge times approaching two seconds to be selected. With this inflow, the pump can be slowed to less than 0.5 cycle per second.

If the cabin usage rate is +01 7.560 kg/sec (1.25 lbm/hr), and the motor gas usage is set to match it, the time to charge an oxygen tank and the amount of oxygen vented into the cabin is shown in Table 2-1 below:

TABLE 2-1

TANK CHARGE TIME AND QUANTITY OF OXYGEN VENTED TO CABIN
VS FINAL TANK PRESSURE

Final Oxygen Tank Pressure	Time to Charge Oxygen Tank, min	Quantity of Oxygen Vented Into Cabin
+07 1.379 Pa (2000 psia)	17	-01 1.633 kg (0.36 lbm)
+07 2.069 Pa (3000 psia)	26	-01 2.495 kg (0.55 lbm)
+07 2.758 Pa (4000 psia)	34	-01 3.221 kg (0.71 lbm)

3. OXYGEN TANK SURVEY

3.1 INTRODUCTION

AiResearch has conducted a survey of high pressure tank manufacturers to determine the availability of off-the-shelf tanks. The survey was limited to cryoformed stainless steel, filament-wound, and Inconel 718 tanks. Cryoformed tanks are fabricated by ARDE, Inc., Mahwah, N.J; filament-wound tanks are fabricated by Brunswick, Inc., Lincoln, Nebraska, and Structural Composities Industries, Azusa, California; and Inconel 718 tanks are fabricated by AiResearch.

3.2 DISCUSSION

3.2.1 Design Criteria

Tentative tank design criteria were established as follows:

Pressure range:	+07 1.379 Pa to +07 4.826 Pa (2000 to 7000 psig)
Usable gas:	-01 7.258 Kg down to +06 1.034 Pa (1.6 ibm down to 150 psia)
Service:	Oxygen gas
Temp. limits:	+02 2.193 t_K to +02 3.443 t_K (-65 to +160°F)
Press. cycles:	10,000 minimum
Proof press:	1.5 x operating
Burst press:	2.0 x operating

Calculations were made to determine tank volume required to hold -01 7.258 kg (1.6 lbm) of usable oxygen at various tank pressures. For these calculations, it was considered that the tank usable oxygen supply was depleted when tank pressure was less than +06 1.034 Pa (150 psia). Results have been plotted in Figure 3-1. Volumes given are independent of tank shape.

Analysis of Figure 3-1 indicates that the volume decrease from +07 2.758 Pa to +07 4.826 Pa (4000 to 7000 psia) is only +04 6.880 m^3 (42 in.^{3}), which is very small when compared with a typical life support system volume of -02 6.556 m^3 (4000 in.^{3}). The most significant volume changes occur in the +06 6.895 to +07 2.068 Pa (1000 to 3000 psia) range. At +06 6.205 Pa (900 psia), which is the spacecraft supercritical cryogenic oxygen supply pressure, oxygen density is +01 8.422 kg/m^3 (3.0426 x 10^{-3} lbm/in.^{3}). At this pressure, a -02 1.028 m^3 (627 in.^{3}) tank would provide the desired -01 7.258 kg (1.6 lbm) of usable oxygen. Since the boost pump would not be needed at pressures below +06 6.205 Pa (900 psia), tank volumes larger than -02 1.028 m^3 (627 in.^{3}) were not seriously considered for the tank survey.

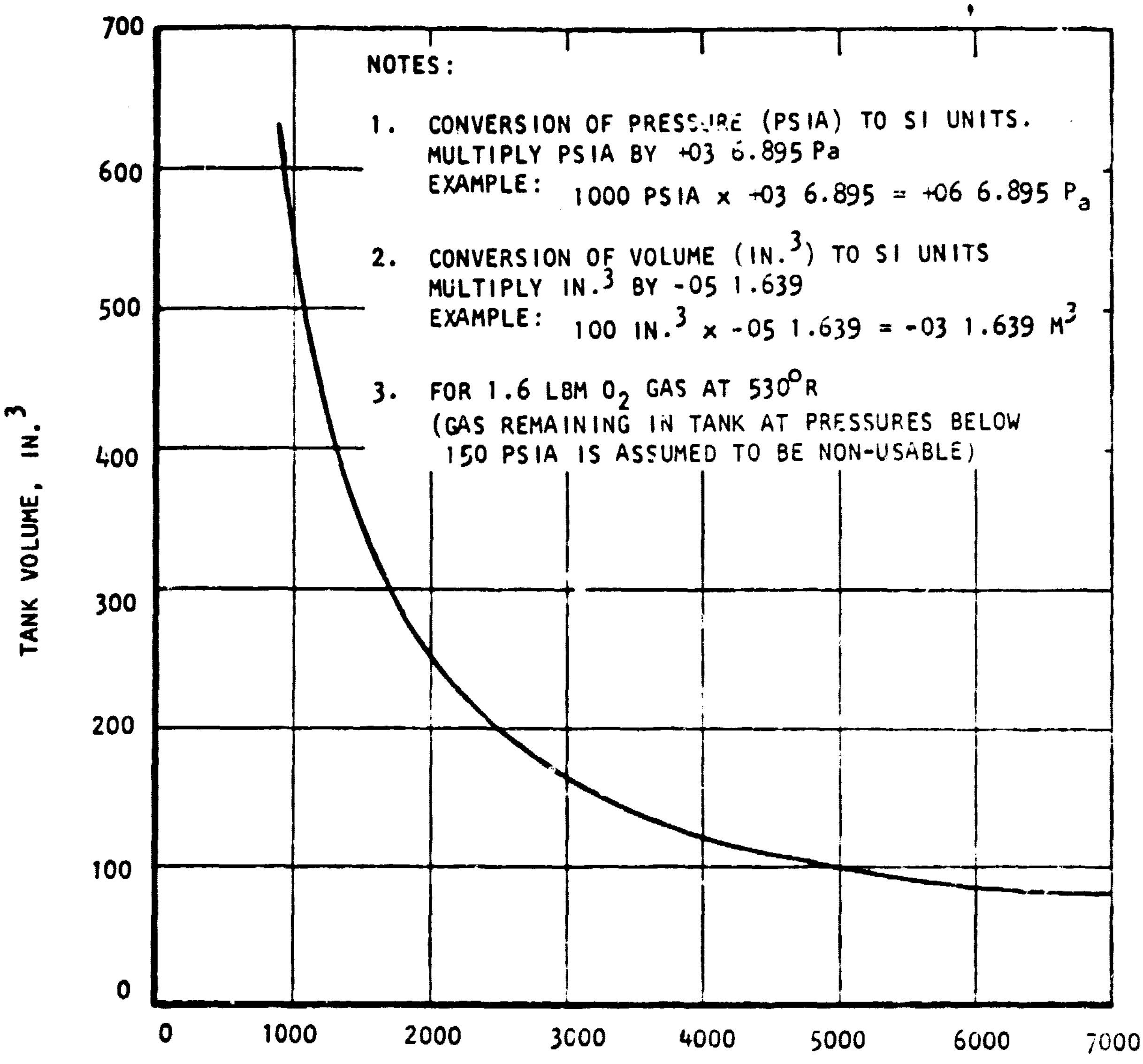

Figure 3-1. Tank Volume vs. Tank Pressure Needed to Provide 1.6 Lbm Usable O_2 Gas

S-1206

3.2.2 Existing Tankage Evaluation

Using the above criteria as a guide, Table 3-1, which is based upon inputs from tank fabricators, was prepared. Information presented includes manufacturer's name, tank shape, volume, size, material, empty weight, maximum design pressure, pressure required to provide -01 7.258 kg (1.6 lbm) of usable oxygen, and previous usage. The tanks listed came closest to meeting the majority of the above noted criteria; however each had some characteristic or combination of characteristics, which made it unacceptable for consideration. Of primary concern was volume, size, weight, and operating pressure.

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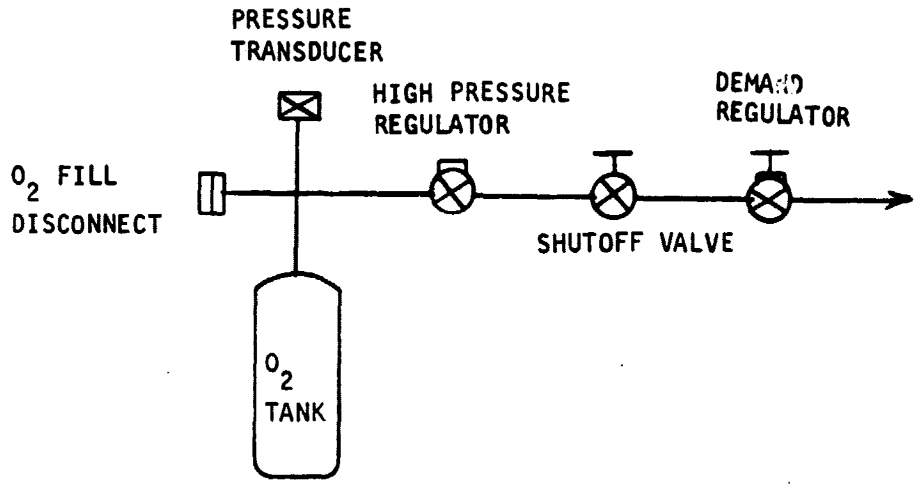
TABLE 3-1

TANK SURVEY SUMMARY

Manufacturer	Shape	Volume, in^3	Size, in.	Material	Empty Weight, lb	Maximum Design Operating Pressure, psia	Pressure Required to Provide 1.6 lbs Usable O_2, psia	Previous Usage
ARDE	Sphere	200	7.5 Dia.	301 SS Cryoformed	4.5	5000	2450	Lockheed Agena Fuel Cell
ARDE	Cylinder	346	6.75 dia. x 15 length	301 SS Cryoformed	14.0	7500	1500	Manned Maneuverability Unit-Gemini
ARDE	Cylinder	378	6.08 dia. x 16 length	301 SS Cryoformed	3.5	1110	1380	Apollo Back Pack (PLSS)
AiResearch	Cylinder	115	4.25 dia. x 13 length	Inconel 718	9.2	6850	4200	Skylab ALSA Secondary Oxygen Pack (SOP)
AiResearch	Cylinder	118	4 dia. x 14 length	Inconel 718	9.8	7500	4140	Gemini Chest Pack
AiResearch	Sphere	76	5.6 dia.	Inconel 718	4.5	7500	7000	Portable Environmental Control System (PECS)
Brunswick	--	938	--	Aluminum liner, filament wound	12.0	5000	<1000	---
Structural Composites Ind.	Cylinder	280	5 dia. x 20 length	Aluminum liner, filament wound	8.5	4000	1800	Proposed

3.2.3 Impact on Other System Hardware

Prior to selection of an optimum tank pressure, it is necessary to evaluate the impact of this pressure on other life support system high pressure components. A typical system schematic for the high pressure circuit is shown below.



In this schematic, the high pressure regulator and all hardware upstream would be subjected to tank pressure. This includes the oxygen fill disconnect, pressure transducer, tubing, and regulator. Each item is evaluated in the following paragraphs.

3.2.3.1 Oxygen Fill Disconnect

Disconnects are normally available up to +07 2.068 Pa (3000 psia) maximum operating pressure. For pressures above +07 2.068 Pa (3000 psia), less are available. There is generally little weight difference between those designed for the lower and higher pressures. Costs for the higher pressure units may be slightly higher and leakage problems are more severe, resulting in lower reliability.

3.2.3.2 Pressure Transducer

Transducers in the pressure range up to +07 1.724 Pa (2500 psia) are readily available. Above this pressure, they are less available. A transducer designed for +07 3.447 Pa (5000 psia) may weigh 5 to 10 percent more than the +07 1.724 Pa (2500 psia) unit and may be 20 to 50 percent more costly.

3.2.3.3 Tubing

The pressure capability of tubing varies with wall thickness, outside diameter, and material. The impact of wall thickness is shown below for -03 6.350 m (0.25 in.) outside diameter stainless steel tubing.

Wall thickness	Maximum operating pressure
-04 8.890 m (0.035 in.)	+07 2.758 Pa (4000 psia)
-03 1.245 m (0.049 in.)	+07 3.447 Pa (5000 psia)
-03 1.651 m (0.065 in.)	+07 4.723 Pa (6850 psia)
-03 1.981 m (0.078 in.)	+07 6.895 Pa (greater than 10,000 psia)

For example, the ALSA Secondary Oxygen Pack, which was designed for +07 4.723 Pa (6850 psia) maximum operating pressure, required two high pressure tube assemblies of -03 1.651 m (0.065 in.) wall thickness. Their total weight was -01 3.375 kg (0.774 lbm). This represents approximately 8 percent of the oxygen tank weight. A weight comparison for various wall thicknesses of -03 6.350 m (0.25 in.) outside diameter stainless tubing is:

Wall Thickness	Weight
-04 8.890 m (0.035 in.)	-01 1.225 kg/ m (0.00686 lbm/in.)
-03 1.245 m (0.049 in.)	-01 1.584 kg/ m (0.00887 lbm/in.)
-03 1.651 m (0.065 in.)	-01 1.941 kg/ m (0.01086 lbm/in.)
-03 1.981 m (0.078 in.)	-01 2.152 kg/ m (0.01205 lbm/in.)

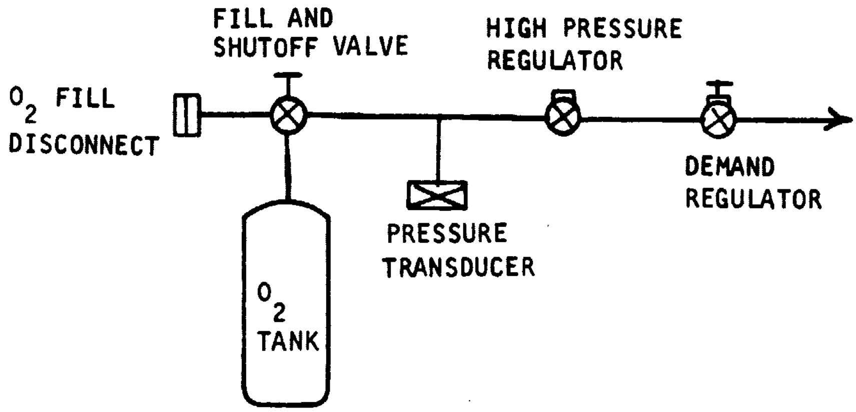
If tubing of -04 8.890 m (0.035 in.) or -03 1.245 m (0.049 in.) wall thickness could have been used, a total tubing weight saving of 36 and 18 percent, respectively, could have been made. There is little difference in material cost between the first three listed wall thicknesses. The last one, -03 1.981 m (0.078 in.), is rated as superpressure tubing and can cost considerably more. Also to be considered are costs associated with preparation of tube ends, bend radii, types of end fittings, etc. The -03 1.981 m (0.078 in.) wall tubing, for example, requires special fittings, which add additional weight and cost.

3.2.3.4 Regulators

Pressure regulators in the range up to +07 2.068 Pa (3000 psia) are also readily available. For units above +07 2.068 Pa (3000 psia), units are less available and may be considered a special design. The weight penalty is slight; however hardware and test costs can be 200 to 300 percent greater for a +07 3.447 to +07 4.826 Pa (5000-7000 psia) regulator. In some applications, higher pressures require two-stage regulators.

3.2.3.5 Alternate Schematic

An alternate schematic is shown below. In this design, the shutoff valve has been moved upstream of the regulator and has been modified to a 3-way fill and shutoff valve. This schematic is typical of a system designed for +07 3.447 to +07 4.826 Pa (5000-7000 psia).



The fill and shutoff valve provides the means for filling the oxygen tank, and tank isolation following filling and during storage. This minimizes potential leakage through the fill disconnect and high pressure regulator. A typical fill and shutoff valve used in the illustrated system can weigh up to -01 9.072 kg (2.lbm). The same valve, if designed for +07 2.068 Pa (3000 psia) or less, would weigh approximately 25 to 35 percent of the high pressure valve weight. Fabrication costs for the high pressure valve are extremely high, seat materials and finishes are very critical, seat coinage (caused by repeated valve operation) is common, and test costs are usually high. All these considerations are minimized at pressures of +07 2.068 Pa (3000 psia) and below.

3.2.3.6 Additional Benefits

Additional benefits that can be expected at lower pressures include: 1) lower design costs, 2) increased safety, 3) greater reliability, 4) improved maintainability, 5) longer life, and 6) lower spares costs.

4. OXYGEN BOOST PUMP DESIGN AND ANALYSIS

4.1 INTRODUCTION

The breadboard boost pump, built and tested under a study program and described in AiResearch Report 74-410521, Rev. A, demonstrated the need for better energy management. Improvements in this area were achieved through improved computer modeling and new mechanical designs. Design innovations include a redesign of the automatic cycle valve with the attendant capacitor, pressure feedback, and control orifices. Adoption of the innovations results in better performance and reduced size, weight, and volume of the boost pump.

In addition, an evaluation of compression thermal effects indicated that sufficient heat dissipation will be inherent to the boost pump design so that the compressed gas temperature will not exceed +02 3.441 t_k (160°F) when operating in cabin ambient temperatures up to +02 2.998 t_k (80°F).

Initial boost pump analysis indicates that the final tank pressure should be limited to +07 2.758 Pa (4000 psia) maximum to avoid seal and energy management development problems.

4.2 DISCUSSION

4.2.1 Alternatives Considered

4.2.1.1 Initial Prototype Design

The initial prototype design, shown in Figure 4-1, resulted from the objective of improving the energy effectiveness of a constant motor pressure, free piston compressor or boost pump. Basically, in a commercial boost pump, constant pressure is applied to a large area motor piston, which strokes a small area compressor piston to produce a smaller volume of higher pressure gas. At the end of the compression cycle, the motor piston swept volume (at the same pressure as supplied) is exhausted to ambient without doing any further work.

The objective was to conceive a mechanism that would utilize a greater percentage of the specific energy of the input gas to do work before venting, and thereby use less input gas.

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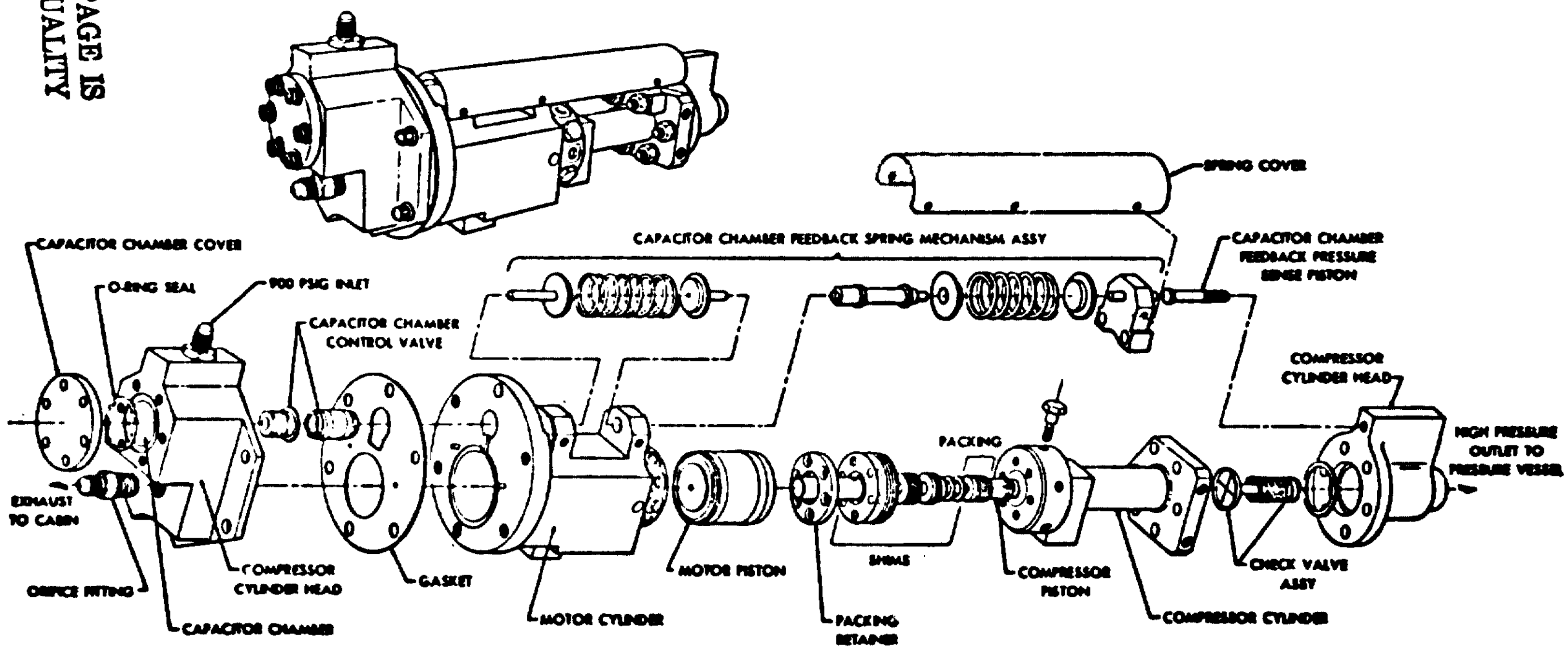


Figure 4-1. Oxygen Boost Pump
Initial Prototype Design

The approach selected was also a free piston, differential area pump, but one that would use the pressure energy of a reduced volume of gas to accelerate the piston mass, while expanding, before the compression load is picked up.

The mechanism selected for accomplishing this objective was an automatic cycling valve. This was a differential area unstable valve with tank pressure feedback reset. When closed, a small area senses pressure in a slowly filling, fixed volume capacitor chamber. When the pressure reaches a predetermined value, the valve opens, exposing a larger area to hold the valve open for capacitor discharge. To supply increasing energy for pumping as tank pressure increases, tank pressure feedback resets or increases the pressure in the capacitor at which the automatic cycling valve cracks to initiate the pumping cycle.

The automatic cycling valve proved to be less than the optimum device from the standpoint of energy management. The pressure drop and time delay through the automatic cycling valve to the motor piston reduced the effective use of available energy.

4.2.1.2 Constant Pressure Discharge

Another approach considered and rejected was to eliminate the mechanical complexity of a variable energy capacitor by pumping at a constant pressure. This could be accomplished by making the compressor exhaust check valve into a fixed pressure relief valve with a setting greater than the maximum tank pressure required. The pump would then compress the gas in the cylinder with no outflow until it reached the relief valve setting. It would then complete the stroke, delivering gas at that pressure. Therefore, the work with each stroke would not change and a fixed volume capacitor would be designed accordingly.

This approach simplifies the mechanization problems, but sacrifices overall cycle efficiency. Pumping efficiency degrades with increasing discharge pressure, and this concept dictates that all work is done at this minimum efficiency, maximum discharge pressure point. It also sacrifices the direct initial precharge of the EVA vessel to source pressure or requires the additional complexity of a bypass precharge circuit.

4.2.1.3 Other Compression Techniques Considered

At initiation of the study and as it progressed, other approaches to both motor and compressor were considered, but were eliminated for reasons given in the following discussion.

a. Vane, gear, screw, and lobe motors and compressors. These devices inherently offer good size and weight-to-power advantages when operated at high speeds, but are not particularly efficient. Additionally, these devices must operate within a narrow speed range to be efficient. Therefore, they lack the flexibility to be able to vary oxygen tank changing time.

In a single stage, unlubricated device, compression ratios are limited to less than three to one, so two stages would be required. Two such devices would be required; one for motor power and another for compression, which would result in greater complexity.

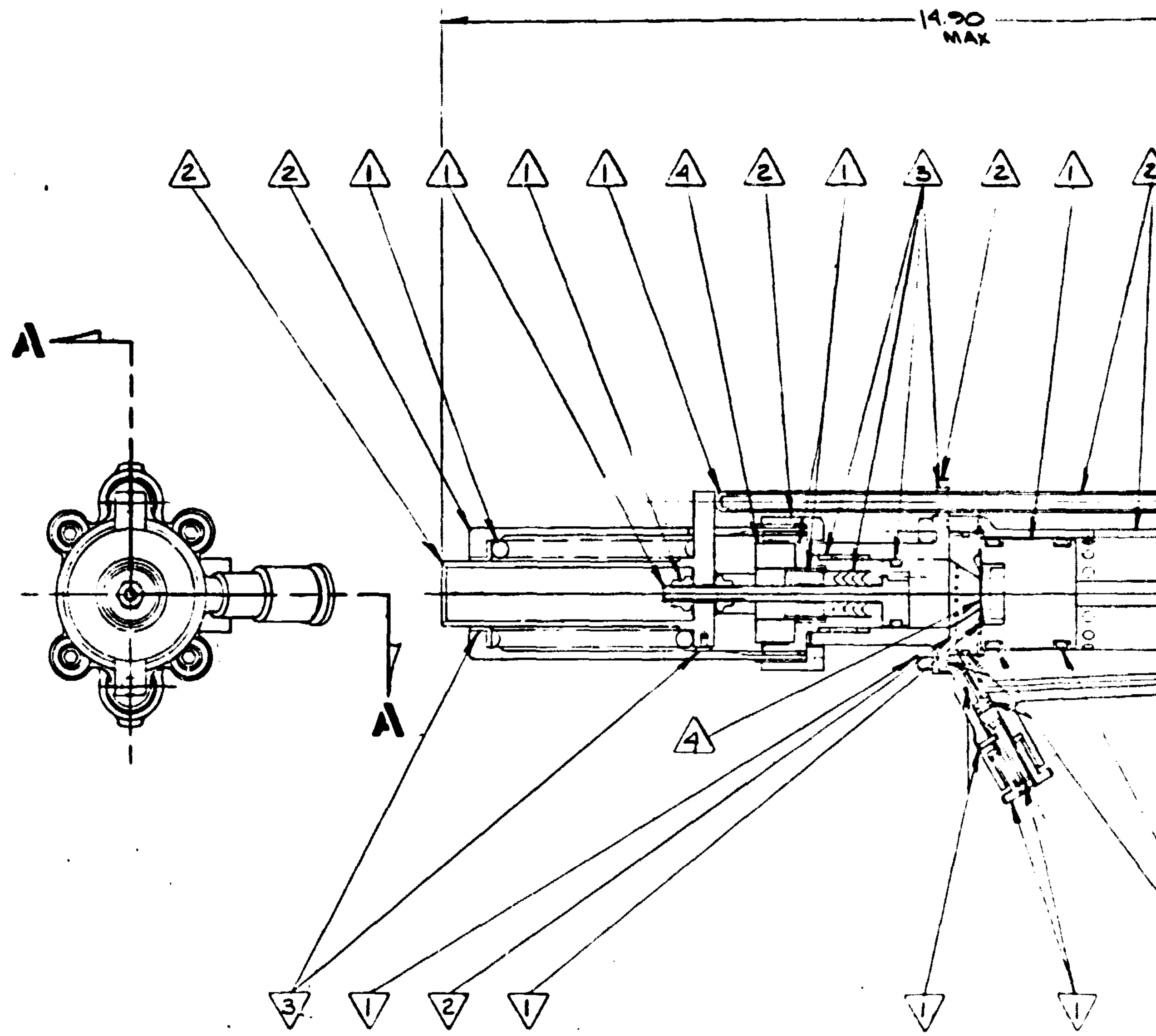
b. Axial and centrifugal compressors. The same comments as above apply. Also, compression ratios attainable for current state-of-the-art designs are even more limited than above.

c. Crank driven reciprocating piston compressors. A design was considered with motor piston(s) and compressor piston(s), mounted on a common crankshaft and valved to provide the desired expansion ratio in motor cylinder(s) and compression ratio in compression cylinder(s), and phased for balance and load distribution. Mechanical complexity would be greater than the selected approach and would be justified only if high flow capacity were a goal. Multi-staging may be required to attain compression ratios greater than 4, to allow intercooling to limit peak temperatures, and to improve volumetric efficiency at high speed when heat transfer within the compression cylinder would not be adequate.

4.2.1.4 Alternatives Considered for the Selected Approach

Prior to the selection of a multiple bellows device to sense pump discharge pressure and schedule capacitor volume, pressure sensing pistons/pushrods were considered. This is shown on Figure 4-2, which follows. A comparison with the selected approach, which is shown on Figure 2-1, Page 2-3, shows the very significant package improvement of the bellows design. By eliminating dynamic seals, which were required on the sensing pistons, the leakage problem was essentially eliminated, as well as friction hysteresis on the accuracy of the scheduling device. The piston-pushrod feedback imposed the inertia and friction of the entire capacitor system on the cyclical action of the capacitor piston. The current bellows feedback mechanism has much less mass and, in addition, only the capacitor piston cycles, independent of the scheduling mechanism. The net effect of the new approach intuitively offers greater efficiency and is reflected in the analysis section of this report.

The bellows design precipitated a reduction in stroke of the scheduling device and, therefore, of the capacitor piston. Metal bellows are typically limited in stroke, so the capacitor diameter was increased to provide the same volume change. Correspondingly, the spring rate increase resulted in shorter springs with fewer convolutions. Then, by stacking two springs in parallel and packaging them within the length generated by the bellows assembly, the length of the overall unit was reduced by more than the total length of the prior spring, even with the addition of the pressure switch and adjusting mechanism.



SECTION A
UNIT SHOWN ENERGISED AT MAXIM

FOLDOUT FRAME /

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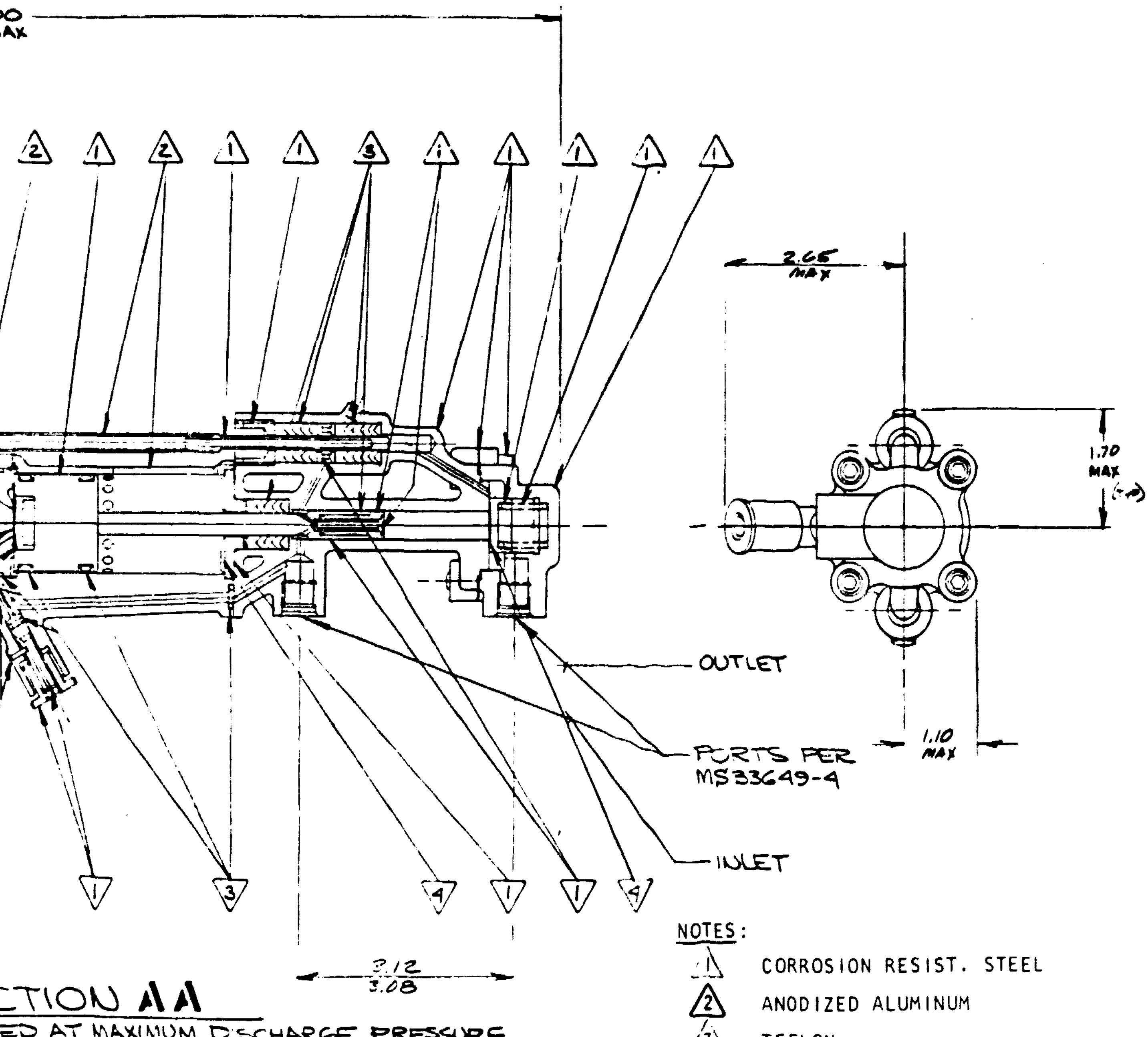

NOTES:

1 CORROSION RESIST. STEEL
2 ANODIZED ALUMINUM
3 TEFLON
4 FLUOROCARBON RUBBER

Figure 4-2. Initial Prototype Pump Design

4.2.2 Description of Selected Flight-Type Prototype

4.2.2.1 General Description

a. Principal Components

1. A compression piston and cylinder, with its associated inlet and outlet valves and seals.
2. A motor piston and seals.
3. A capacitor chamber, piston and spring, trigger port and associated seals.
4. An output pressure sensing bellows assembly and sense tube.
5. A compound thread-driven, adjustable capacity, metering valve.
6. A compressor discharge pressure sensing electrical switch, switch mount and adjustable switch actuator.

b. Design Data

Supply pressure	900 psia oxygen
Compressor discharge pressure	4015 psia max
Motor displacement	1.88 cu in.
Compressor displacement	0.188 cu in.
Motor area	1.5 sq in.
Trigger pressure	645 psia
Pumping frequency at maximum pressure	0.55 Hz
Capacitor volume at 900 psia	0.15 cu in.
Capacitor tank pressure volume	0.3933 cu in. max

4.2.2.2 Detailed Description (Refer to Figure 2-1, page 2-3)

a. Compression Piston. The compression piston contains an integral poppet-type inlet valve, which opens to fill the cylinder during the return or suction stroke. Multiple, teflon, chevron-type seals are retained to the piston by a press fit piston end, which also retains the inlet valve poppet. Integral porting in the piston and in the cylinder routes supply pressure to the inlet valve. A set of multiple, teflon chevron-type seals at one end of the cylinder isolates supply pressure from the back side of the motor piston. The compression piston is not rigidly connected to the motor piston; thus,

loads that could possibly arise due to concentricity tolerances are eliminated. Motor pressure and compressor pressure maintain a force balance that insures synchronization of the two pistons at all times. During the compression stroke, when cylinder pressure exceeds that at the outlet port, the exhaust check valve opens and the remainder of the swept volume is pumped. The compression piston strokes completely to the end of the cylinder, contacting the check valve before it closes to insure an absolute minimum of compressed gas reexpansion for maximum volumetric efficiency. The compressor piston has a diameter of -02 1.014 m (0.399 in.) and a stroke of -02 3.81 m (1.5 in.) for an L/D ratio of 3.75.

Supply pressure +06 6.204 Pa (900 psia) entering the compression chamber through the inlet check valve provides motive force for the return or suction stroke. To insure that the exhaust check valve closes properly, its spring is designed to accelerate the mass of the check valve in the closed direction at a greater rate than the supply pressure can accelerate the mass of the compressor and motor piston. In this way, no compressed gas reenters the compression chamber before the exhaust check valve closes.

b. Motor Piston. The motor piston contains two low friction guide seals and the seat for the trigger. The motor cylinder contains a set of diametral vent holes which are uncovered by the motor piston at the end of the power stroke to vent the power gas to cabin. A second set of diametral vent holes at the end of the motor cylinder vent the back side of the power piston during the power stroke. A fluorocarbon rubber bumper, bonded to the compression piston retainer, cushions the motor piston at end of stroke. A constant flow orifice in the motor piston serves to vent motor pressure during the return stroke. Because the power and return stroke act very fast, little energy is lost through this orifice. The trigger area and the compressor piston area are similar, which allows triggering over a broad range of supply pressures. The motor vents are connected and a silencer can be added, if it proves necessary.

c. Capacitor. The capacitor functions to accumulate energy in the form of gas pressure at a slow rate, as regulated by the metering valve (see Item e). When the force developed by increasing capacitor pressure acting over the trigger area on the motor piston approaches supply pressure acting on the compressor piston area, the trigger valve starts to leak, exposing motor piston area to capacitor pressure, and the motor piston moves. Initial movement fully uncovers the trigger port, which exposes the full motor piston area to capacitor pressure. The capacitor variable volume, which increases with output pressure, is designed to provide the proper amount of energy to just complete the compression stroke. The capacitor piston is pressure balanced, via an orifice through the piston, to reduce the force and size of the capacitor spring. The capacitor piston is shown on Figure 2-1, page 2-3, in its minimum capacitor volume position. During the capacitor charging cycle, flow through the capacitor piston balance orifice insures that differential pressure and force across the capacitor piston is at, or near, zero. A flat

wave spring is used to apply a force on the capacitor piston toward the bellows sufficient to overcome friction, insuring the proper capacitor volume change.

d. Bellows Assembly. The bellows assembly located above the capacitor functions to schedule the position of the capacitor piston to increase capacitor volume as pump output or tank pressure increase. The bellows assembly consists of two concentric bellows, connecting the housing and the actuator rod. The bellows assembly is constructed so that both bellows sense pump output or tank pressure on their outside diameters. The effective area of the assembly is the effective area of the large O.D. bellows, minus the effective area of the small O.D. bellows. The net rate of the bellows and capacitor springs is designed to produce the stroke required of the capacitor piston.

During the capacitor discharge or pumping cycle, which is very fast, the balance orifice permits only limited flow out of the balance chamber, so that the balance chamber acts as a pneumatic spring. The capacitor piston therefore strokes to a minimum capacitor volume position with every pumping cycle. The bellows assembly force balance remains essentially unchanged throughout the charge and pumping cycle. Because it has a very high spring rate, it has only a limited stroke with every pumping cycle. The diameter of the capacitor has been maximized consistent with the motor package to reduce the stroke of the bellows over the tank pressure range, and to reduce cyclical stroke of the capacitor piston during pumping.

e. Capacitor Metering Valve. The capacitor metering valve is designed to enable fine adjustment of flow area to closely control the capacitor charge rate (and, therefore, pumping rate) to match designed oxygen flow into the cabin. When desired, the area can be opened to increase cycle rate to maximum to allow rapid PLSS pumpup. The needle is a scarfed, constant cross section, rather than a tapered needle, which presents a larger cross section for any given flow area, making it less sensitive to microscopic particulate plugging. The needle advance is a compound thread, which allows finer, more positive adjustment.

f. Discharge Pressure Sensing Electrical Switch. This switch provides a discrete signal, off or on, as a function of compressor output/tank pressure to stop the pump at a preselected maximum pressure. The electrical switch is rigidly mounted and is actuated by an adjustable screw. The actuator screw is mounted into the bellows assembly spring retainer. It directly indicates the position or stroke of the bellows assembly, which is a direct function of compressor output pressure.

The electrical switch is Honeywell Part IISM244, which has passed explosive atmosphere testing. The switch has a differential actuator stroke of -04 1.016 m (0.004 in.). The bellows assembly has a differential stroke with each pumping cycle of the same order of magnitude. Since the response of the shutoff device is not known it is recommended that the switch be used to operate a latching relay to prevent on-off oscillations of the shutoff devices to immediately stop the pumping cycle.

4.2.3 Analysis of Flight-Type Prototype

4.2.3.1 Computer Model and Performance Prediction

The analysis of the final boost pump design involved a detailed computer simulation of the device. The simulation consisted of a series of first order differential equations solved numerically on the AiResearch/CDC 6400 digital computer. The numerical integration technique consisted of a "predictor-corrector" method with a variable time increment of integration. The time increment of integration was selected by an iteration scheme which assures accuracy of solution and at the same time allows the integration to proceed at the maximum speed. A discussion of the equations which were solved numerically follows. Refer to Figure 4-3 for identification of symbols used in the equations.

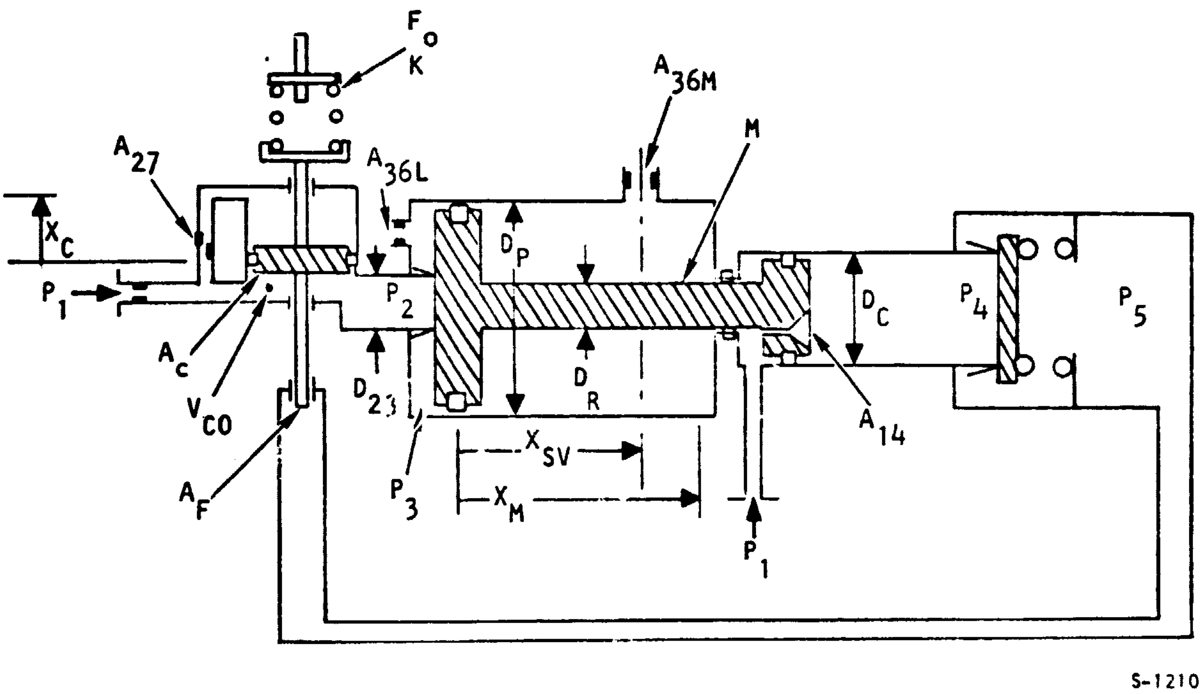

Figure 4-3. Identification of Symbols Used in Equations

Boost pump pressure, P_4, was obtained by integrating the equation:

$$\frac{d\,(P_4\ V_4)}{dt} = RT_4\ \frac{dM_4}{dt}, \qquad (1)$$

which can be written as follows:

$$\frac{d\,(P_4\,V_4)}{dt} = K\,R\,T_4\left(\frac{P_1\,A_{14}\,N_{14}}{\sqrt{T_1}} - \frac{P_4\,A_{45}\,N_{45}}{\sqrt{T_4}}\right), \text{ where:} \tag{2}$$

V_4 = compression chamber volume, in.^{3}
K = nozzle flow factor, $\sqrt{^0R}$/sec
R = gas constant, in/^{0}R
T_4 = compression chamber temperature, ^{0}R
P_1 = supply pressure, psia
A_{14} = inflow valve effective area, in.^{2}
N_{14} = pressure ratio function (depending upon P_1/P_4)
T_1 = supply gas temperature, ^{0}R
A_{45} = tank check valve effective area, in.^{2}
N_{45} = pressure ratio function (depending upon P_4/P_5)
P_5 = tank pressure, psia

It should be noted that A_{14}, the inflow valve effective area, is essentially zero when P_4 is greater than P_1 and that A_{45}, the tank check valve effective area is essentially zero when P_5 is greater than P_4

The compression chamber temperature, T_4, was computed from the equation

$$T_4 = T_1\left(\frac{P_4}{P_1}\right)^{\frac{\gamma-1}{\gamma}}, \quad \text{where:} \tag{3}$$

γ = ratio of specific heats

The equation for T_4 is the equation for adiabatic compression of gas from pressure P_1 to pressure P_4.

After equation (2) was integrated, P_4 was obtained by dividing the compression chamber volume V_4.

The compression chamber volume, V_4 was obtained from the equation:

$$V_4 = A_{P4}\,(X_{MAX} - X), \quad \text{where:} \tag{4}$$

A_{P4} = area of boost pump piston, in.^{2}
X_{MAX} = maximum stroke of boost pump, in.
X = position of boost pump, in.

The position of the boost pump was obtained by integrating the velocity of the piston.

The velocity of the piston was obtained by integrating the unbalanced forces on the piston divided by the mass of the piston.

The terms comprising the unbalanced force on the piston were as follows:

$$P_2 A_T + P_3 (A_P - A_T) - P_6 A_P + P_1 (A_{P4} - A_R) - P_4 AP_4 \pm F_f \qquad (5a)$$

for zero and small piston stroke, and using:

$$P_3 A_P - P_6 A_P + P_1 (A_{P4} - A_R) - P_4 (A_{P4}) \pm F_f \qquad (5b)$$

for large piston strokes, where:

P_2 = capacitor pressure, psia
P_3 = motor chamber pressure, psia
P_6 = ambient pressure, psia
A_P = area of motor piston, in.^{2}
A_T = area of trigger valve, in.^{2}
A_R = area of connecting rod, in.^{2}
F_f = coolant friction, lbf

The equations relating motor chamber pressure, temperature, and volume are presented as follows:

$$\frac{d\,U_3}{dt} = \frac{d\,Q_3}{dt} - \frac{d\,W_3}{dt} + h_{in}\frac{d\,M_{in}}{dt} - h_{out}\frac{d\,M_{out}}{dt}, \quad \text{where:} \qquad (6)$$

U_3 = internal energy of motor chamber
Q_3 = heat transfer to (from) motor chamber
W_3 = work done by motor chamber
h_{in} = enthalpy of gas entering motor chamber
M_{in} = mass of gas entering motor chamber
h_{out} = enthalpy of gas leaving motor chamber
M_{out} = mass of gas leaving motor chamber

Equation (6) can be rewritten, assuming no heat transfer, as follows:

$$\frac{d}{dt}\left(C_v\ T_3\ \frac{P_3\ V_3}{R\ T_3}\right) = \frac{C_p\ T_2\ K\ P_2\ A_{23}\ N_{23}}{\sqrt{T_2}} - \frac{C_p\ T_3\ K\ P_3\ A_{36}\ N_{36}}{\sqrt{T_3}} - (P_3 - P_6)\ A_p\ U \tag{7}$$

where:

C_v = specific heat of gas at constant volume

C_p = specific heat of gas at constant pressure

A_{23} = effective flow area of trigger valve

N_{23} = pressure ratio function (depending upon P_2/P_3)

A_{36} = motor chamber vent effective area (note that A_{36} is minimum for piston strokes less than X_{SV} and maximum for strokes greater than X_{SV})

X_{SV} = position of motor chamber vent valve

N_{36} = pressure ratio function (depending upon P_3/P_6)

U = velocity of piston

By utilizing the relations:

$$\frac{C_p}{C_v} = \gamma \text{ and } \frac{R}{C_v} = \gamma - 1$$

Equation (7) was further simplified into the form in which it was integrated.

$$\frac{d}{dt}(P_3\ V_3) = (K\ R\sqrt{T_2}\ P_2\ A_{23}\ N_{23} - K\ R\sqrt{T_3}\ P_3\ A_{36}\ N_{36})\ \gamma - (P_3 - P_6)\ A_p\ U\ (\gamma - 1) \tag{8}$$

The continuity equation for the motor chamber can be written as follows:

$$\frac{d}{dt}\left(\frac{P_3\ V_3}{T_3}\right) = K\ R\left(\frac{P_2\ A_{23}\ N_{23}}{\sqrt{T_2}} - \frac{P_3\ A_{36}\ N_{36}}{\sqrt{T_3}}\right) \tag{9}$$

The motor chamber volume, V_3, was computed from the equation:

$$V_3 = V_{30} + A_p\ X, \quad \text{where:} \tag{10}$$

V_{30} = dead volume of motor chamber, in.^3

P_3 was obtained by integrating equation (8) and then dividing by V_3. Motor chamber temperature, T_3 was obtained by multiplying P_3 times V_3 and then dividing by the integral of equation (9).

Capacitor pressure, P_2, was obtained by integrating the equation:

$$\frac{dP_2}{dt} = \frac{K R \sqrt{T_4}}{V_c} P_2 \left(A_{12} C_{12} - A_{27} N_{27} - A_{23} N_{23} \right) - \frac{P_2}{V_c} \left(A_c U_c \right), \text{ where:} \quad (11)$$

V_c = volume of capacitor chamber, in.^{3}

A_{12} = eff. area of capacitor inflow valve, in.^{2}

C_{12} = pressure ratio function (depending upon P_1/P_2)

A_{27} = capacitor balance chamber orifice effect area, in.^{2}

N_{27} = pressure ratio function (depending upon P_2/P_7)

P_7 = capacitor balance pressure, psia

A_c = area of capacitor piston, in.^{2}

U_c = velocity of capacitor piston, in./sec

The volume of the capacitor was computed from the equation:

$$V_c = V_{co} + X_c A_c, \quad \text{where:} \quad (12)$$

V_{co} = dead volume of capacitor, in.^{3}

X_c = position of capacitor piston, in.

The position of the capacitor piston was calculated by integrating the capacitor piston velocity. The capacitor piston velocity was obtained by integrating the unbalanced force on the capacitor piston. The unbalanced force on the capacitor piston was comprised of the following terms.

$$P_5 A_F + (P_c - P_7) A_c - F_{SO} - K_c X_c, \quad \text{where:} \quad (13)$$

P_5 = tank pressure (feedback)

A_F = tank pressure feedback area, in.^{2}

F_{SO} = pre-load in capacitor spring, lbf

K_c = spring rate of capacitor spring, lb/in.

The capacitor balance pressure, P_7, was obtained by integrating the equation:

$$\frac{d P_7}{dt} = \frac{K R\sqrt{T_1}}{V_7} \left(P_7 A_{27} C_{27} \right) + \frac{P_7}{V_7} A_c U_c \quad (14)$$

where V_7 = volume of capacitor balance chamber, in.^{3}

V_7 was obtained from the equation:

$$V_7 = V_{70} - A_c X_c, \text{where:} \quad (15)$$

V_{70} = volume of balance chamber at zero stroke

4.2.3.1.1 Predicted Performance at 900 psia Supply

The gas compressed per cycle of operation was:

$$\frac{(914.7)(0.125)(1.5)}{(580)(530)} = 0.000558 \text{ lbm}$$

With a cabin flow rate of 3.60 lbm/hour, the boost pump cycled at rates shown in Figure 4-4.

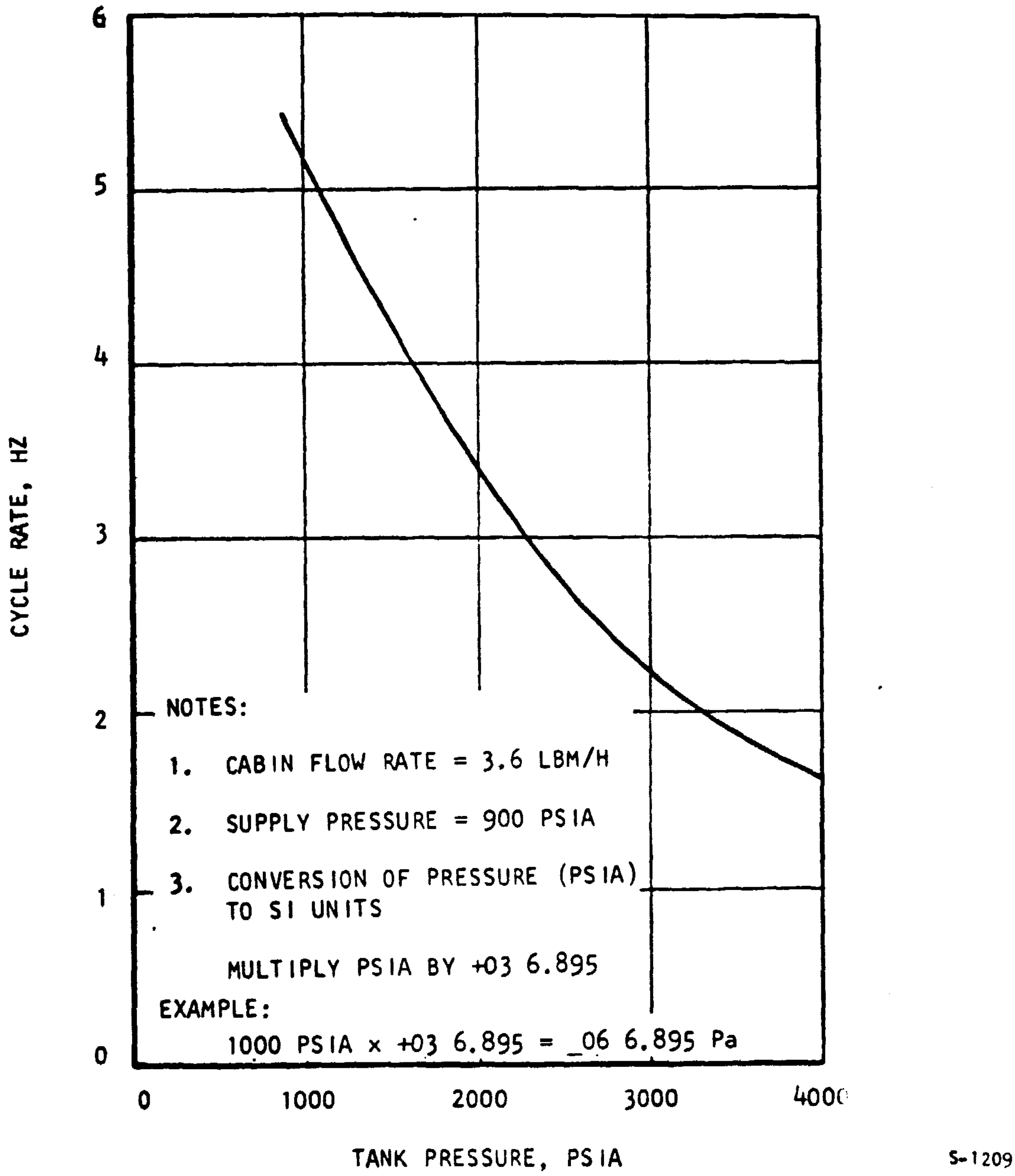

Figure 4-4. Cycling Rate for 3.6 Lbm Hr Cabin Flow Rate

From Figure 4-4, the data in Table 4-I was extracted by graphical means.

TABLE 4-I

AVERAGE CYCLE RATE VS.
FINAL TANK PRESSURE FOR
CABIN FLOW = 3.6 LB/HR
INITIAL PRESS = 900 PSIA

Final Tank Pressure Pa	(psia)	Cycle Rate (Hz)
+06 6.895	1000	5.28
+07 1.379	2000	4.37
+07 2.069	3000	3.65
+07 2.758	4000	3.10

The total gas flow into the cabin required to store 1.6 lbm at various final tank pressures was computed according to the following procedures:

1. Compute tank volume.

   $P_F V = 1.6\ RT$

   $V = 491840/P_F$, where: $V = \text{in.}^3$, $P_F = \text{psia}$

2. Compute compressed gas.

   $\Delta M_{COM} = 1.6 - M_I$

   M_I = initial mass of gas in tank = $\dfrac{(914.7)(V)}{R\ T} = 0.00297\ V$

   Thus, $\Delta M_{COM} = 1.6 - 0.00297\ V$

3. Compute time to compress gas at 3.6 lb/hr flow rate.

   $$\Delta t = \frac{\Delta M_{COM}}{f_A\ (0.000558)}$$

4. Compute supply flow = $\Delta M_S = \Delta t\ (3.6)$

Figure 4-5 shows graphically the predicted compressor performance in terms of supply gas versus final tank pressure for an initial pressure of 900 psia and a total compressed mass of 1.6 lbm for both the initial and final flight weight prototype designs.

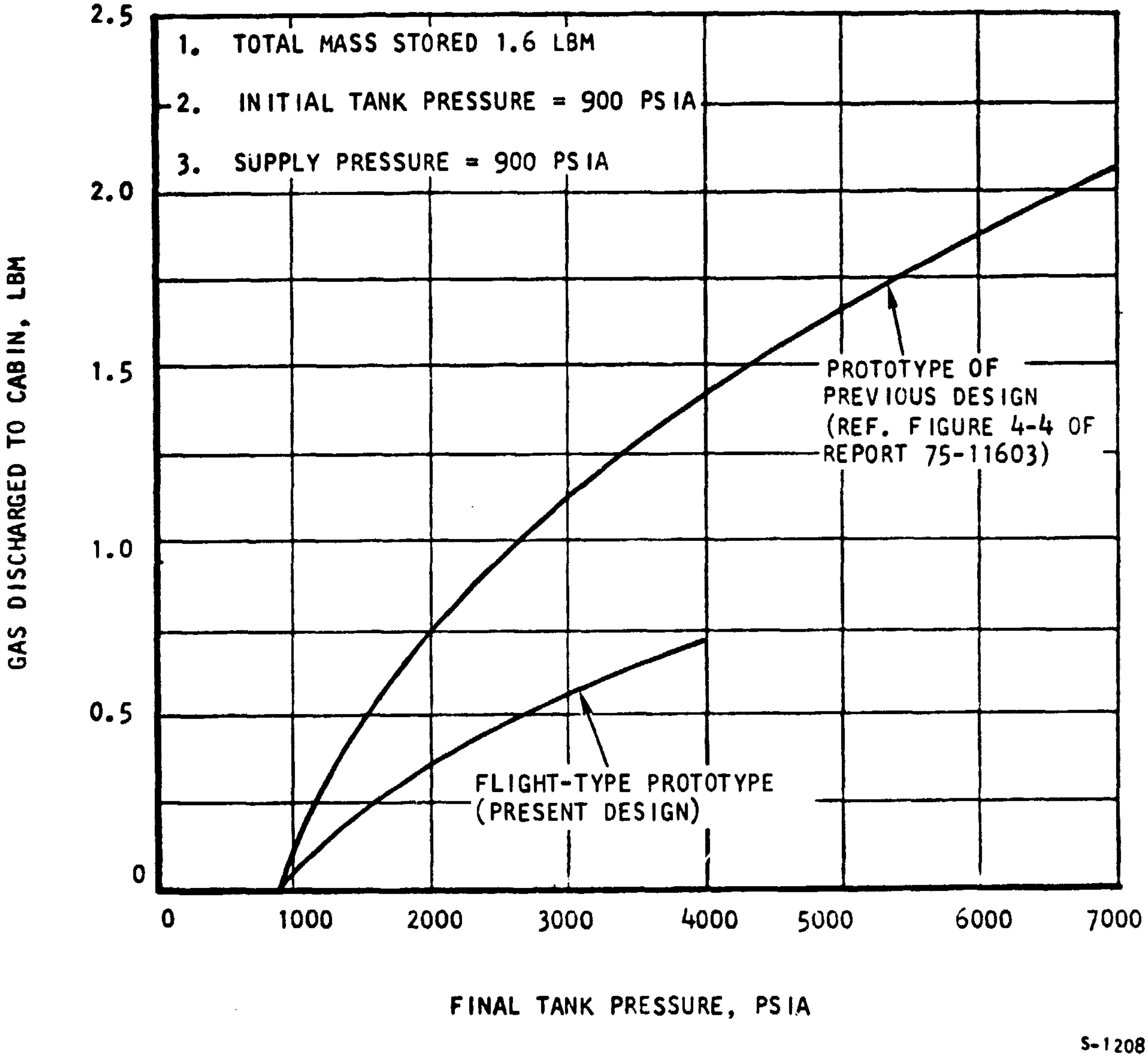

Figure 4-5. Oxygen Boost Pump Predicted Performance for Previous and Present Designs

It should be noted that the performance shown in Figure 4-5 is considerably better than was predicted in AiResearch Report 75-11603, Figure 4-4. This led to a comparison of the performance of the two boost pumps. The boost pump discussed in 75-11603, referred to herein as the "previous" design, was of a larger displacement than the present boost pump. The frequency of oscillation of the two boost pumps was normalized by multiplying by the mass of gas compressed during each cycle and dividing by the cabin flow rate. Expressed mathematically,

$$\bar{f} = \frac{f\ M_0}{W_c}, \text{ where:}$$

$\bar{f}$ = non-dimensional frequency

f = actual frequency of oscillation (Hz)

M_0 = mass of gas displaced during one cycle (lbm)

W_c = cabin flow rate (lbm/s)

A comparison of $\bar{f}$ vs tank pressure is shown on Figure 4-6. It is seen that current design oscillates faster at low tank pressures. This was found to result from a more optimum capacitor volume scheduling for the present design than for the previous design. At low pressures, the previous design capacitor volume was excessive, resulting in slow cycle rates and excessive compressor speed at end of stroke. At higher tank pressure, the capacitor volume scheduling is considered to be more nearly optimum, as evidenced by the similarity of the non-dimensionalized frequency curves. For the present design, the capacitor volume vs tank pressure schedule is considered to be optimum for the range of tank pressures of 900 to 4000 psia.

At tank pressures above 4000 psia, the compressor continues to operate with incomplete stroking until approximately 9000 psia is reached, at which time the boost pump continues oscillating, but does not achieve displacement of gas into the tank.

4.2.3.1.2 Predicted Performance at Low Supply Pressure

The predicted performance of the boost pump at a supply pressure of 500 psia is presented in Figure 4-7. The performance is presented in terms of non-dimensional frequency $\bar{f}$ (as discussed in paragraph 4.2.3.1.1). Performance for three different conditions is shown. Performance at 900 psia is included as a reference and is identical to that shown in Figure 4-6. 500 psia performance is shown for two conditions. One condition is for the same capacitor scheduling mechanism as was used for the 900 psia supply performance. The other condition is for a capacitor scheduling mechanism which reaches twice the volume at a given tank pressure, as did the original scheduling mechanism. The 500 psia data for both scheduling mechanisms is significantly degraded from that obtained at 900 psia. Two phenomena appear to account for the difference between operation at 500 psia and 900 psia:

1. Non-optimum capacitor scheduling for the 500 psia cases.

2. Increased overall compression ratio for the 500 psia cases.

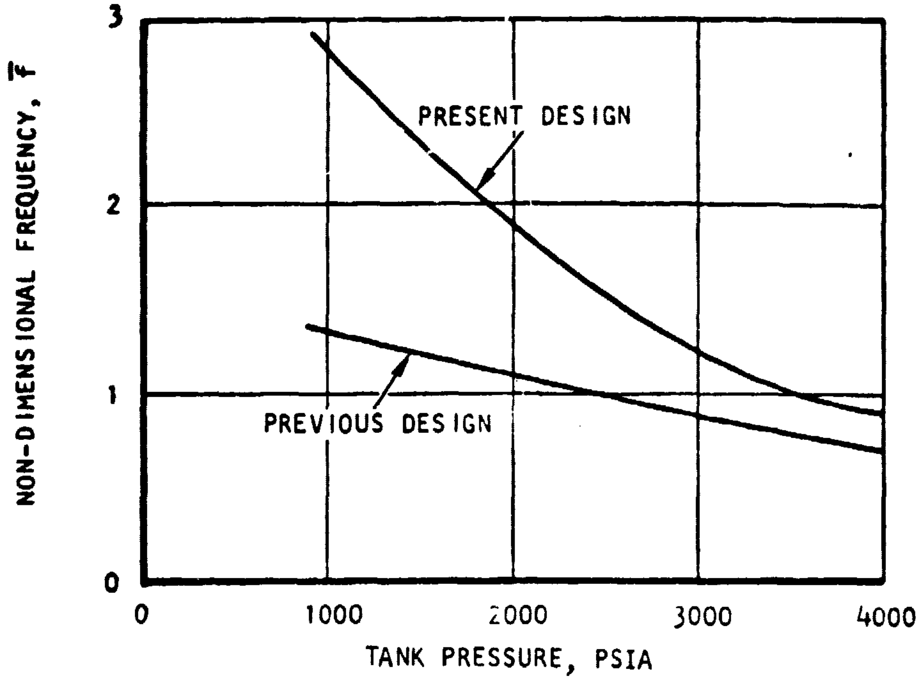

Figure 4-6. Non-Dimensional Frequency for Previous and Present Designs

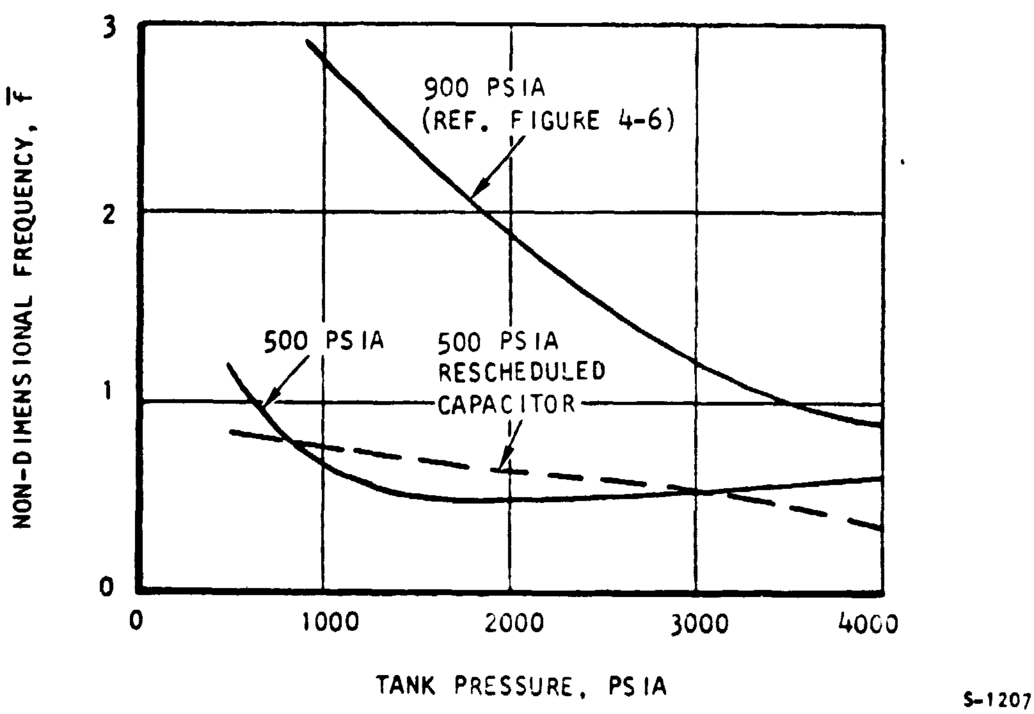

Figure 4-7. Non-Dimensional Frequency for 500 psia Supply Pressure

4.2.3.1.3 Sizing Variations

During the course of analysis of the O_2 boost pump, various size compression chambers were considered. The variation was considered to result both from length and diameter changes. It was concluded that outside of heat transfer and mechanical design considerations, performance, that is mass compressed per mass flowed into the cabin, was not affected by the boost pump sizing, providing that the capacitor volume vs tank pressure schedule was properly designed.

4.2.3.1.4 Program Changes from 74-410521A

The major computer program change made from that which was described in AiResearch Report 74-410521A consisted of eliminating the automatic cycle valve and replacing it with the balanced piston variable capacitor volume. Also, the flow restrictor between the automatic cycle valve and the motor chamber was removed.